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# Effect of External Pressure Drop on Loop Heat Pipe Operating Temperature


Jentung Ku, Laura Ottenstein
NASA Goddard Space Flight Center
Greenbelt, Maryland, USA
(301) 286-3130
jentung.ku@gsfc.nasa.gov

Paul Rogers
U.S. Army TACOM
Warren, Michigan, USA

Kwok Cheung
Naval Research Laboratory
Washington, D.C., USA



**Abstract**

This paper discusses the effect of the pressure drop on the operating temperature in a loop heat pipe (LHP). Because the evaporator and the compensation chamber (CC) both contain two-phase fluid, a thermodynamic constraint exists between the temperature difference and the pressure drop for these two components. As the pressure drop increases, so will the temperature difference. The temperature difference in turn causes an increase of the heat leak from the evaporator to the CC, resulting in a higher CC temperature. Furthermore, the heat leak strongly depends on the vapor void fraction inside the evaporator core. Tests were conducted by installing a valve on the vapor line so as to vary the pressure drop, and by charging the LHP with various amounts of fluid. Test results verify that the LHP operating temperature increases with an increasing differential pressure, and the temperature increase is a strong function of the fluid inventory in the loop.


## Introduction

Loop Heat Pipes (LHPs) are versatile two-phase heat transfer devices that have gained increasing acceptance for spacecraft thermal control. As shown in Figure 1, an LHP consists of an evaporator, a compensation chamber (CC), a condenser, and vapor and liquid lines. Detail descriptions of the LHP construction and its operating characteristics can be found in the literature [1, 2].

Because both the evaporator and the CC contain two-phase fluid, there exists a thermodynamic relation that governs the pressure drop and the temperature drop between these two elements. As the pressure drop increases, so will the temperature difference. An increasing temperature difference leads to a higher heat leak from the evaporator to the CC and causes the CC temperature to increase. Furthermore, the heat leak strongly depends on the vapor void fraction inside the evaporator core. The higher the vapor void fraction, the higher the heat leak. In ground testing, the vapor void fraction is a function of the fluid inventory in the loop and the relative tilt between the evaporator and the CC.

This paper presents a study of the effect of the pressure drop on the LHP operating temperature. A theoretical background will be presented first, followed by a description of the test article and test set-up. Test results under various conditions will then be described in detail. The implication and relevance of the results to LHP ground testing will also be addressed.

## Theoretical Background

The heat load applied to the evaporator can be divided into two parts. The first part is used to evaporate the liquid and form the flow circulation around the loop. The second part is transmitted to the CC and is commonly referred to as the heat leak. It can be expressed as:

$$Q_e = Q_{e,cc} + Q_{e,vap} \quad (1)$$
$$Q_{e,vap} = m\lambda \quad (2)$$
$$Q_{e,cc} = G_{e,cc} (T_e - T_{cc}) \quad (3)$$

where
$Q_e$ = heat load applied to the evaporator,
$Q_{e,cc}$ = heat leak to the CC,
$Q_{e,vap}$ = amount of heat used to vaporize the liquid,
m = mass flow rate,
$\lambda$ = heat of vaporization of the working fluid, and
$G_{e,cc}$ = thermal conductance between the evaporator and CC.

The saturation temperature of the CC determines the loop operating temperature. The CC can exchange energy with the environment, the evaporator, and the liquid returning from the condenser. For a well-insulated CC, the heat leak is balanced by the subcooling of the returning liquid during steady state,

$$Q_{e,cc} = mC_p \,\Delta T = mC_p (T_{cc} - T_{in}) \quad (4)$$

where $C_p$ is the liquid specific heat, $\Delta T$ is the amount of liquid subcooling, $T_{cc}$ is the CC temperature, and $T_{in}$ is the liquid temperature at the entrance to the CC. The liquid exiting the condenser will exchange heat with its surroundings as it flows along the liquid line. The temperature difference can be expressed as:

$$T_{in} - T_{cond} = Q_{l,a} / (mC_p) \quad (5)$$

where $T_{cond}$ is the temperature of the liquid leaving the condenser, and $Q_{l,a}$ is the heat leak to the liquid from the surroundings. At a low mass flow rate, the liquid will be heated to near the ambient temperature. As the mass flow rate increases, the inlet temperature $T_{in}$ becomes more subcooled. Thus, the inlet temperature is a function of the evaporator heat load, the condenser sink temperature, and the ambient temperature.

Because the CC and the evaporator outer grooves both contain two-phase fluid, a relationship exists between the temperature difference and the pressure difference as expressed by the Clausius-Clapeyron equation:

$$\Delta P = \lambda \,\Delta T / (T_{cc} \,\Delta v) = \lambda (T_e - T_{cc}) / (T_{cc} \,\Delta v) \quad (6)$$

where $\Delta v$ is the difference in the vapor and liquid specific volumes, $\Delta P$ is equal to the total system pressure drop minus the pressure drop through the wick, and $\Delta T$ is the temperature difference between the evaporator and the CC. As the pressure drop increases, so will the temperature difference (for given heat load, sink temperature and ambient temperature). Such a higher pressure drop could result from a larger elevation between the evaporator and the condenser in ground testing. Figure 2 illustrates how these temperatures change with an increasing pressure drop. Initially, the CC and the evaporator are at thermodynamic states of ($P_{cc,1}$, $T_{cc,1}$) and ($P_{e,1}$, $T_{e,1}$), respectively. When a higher pressure drop is imposed, the evaporator temperature will increase to $T_{e,1}^*$. However, the resulting higher temperature difference leads to a higher heat leak and causes the CC temperature to increase. Thus, both $T_{cc,1}$ and $T_{e,1}$ will move along the saturation line. At the new steady state, the temperature difference will match the corresponding pressure drop and the Clausius-Clapeyron relation will be satisfied at the new CC temperature, $T_{cc,2}$.

The thermal conductance $G_{e,cc}$ in Equation (3) is highly dependent upon the vapor void fraction inside the evaporator core. When the evaporator core is completely filled with liquid, heat transfer between the CC and the evaporator is by heat conduction through the hermetic case, and $G_{e,cc}$ is usually small. However, if vapor is present in the liquid core, additional heat is transmitted by conduction through the primary wick and then by evaporation and condensation through the vapor arteries inside the secondary wick, much like in a heat pipe [2, 3]. Such a vapor connection almost always exists and is the primary heat transfer mechanism between the CC and the evaporator. In ground testing, the fluid will form a stratified flow inside the CC and the evaporator core. Thus, the void fraction will be dependent upon the fluid inventory and the relative tilt between the evaporator and CC. The net result is that the loop operating temperature will change with the imposed external pressure drop, and this effect will be highly dependent upon the fluid inventory and the tilt in ground testing. Furthermore, such an effect is more pronounced at low heat loads. As the heat load increases, the higher mass flow rate can provide additional subcooling to partially compensate for the increasing heat leak.

**Test Article and Test Set-up**

Figure 3 shows a schematic of the test article. The evaporator and the CC are made of stainless steel with an O.D. of 25.4 mm, and lengths of 30.5 mm and 12.7 mm, respectively. The primary wick is made of sintered powder nickel with a pore size of 1.3 microns and a permeability of $1.3 \times 10^{-14}$ $m^2$. The vapor line has an I.D. of 3.34 mm and a length of 1.94m while the liquid line has an I.D. of 1.75 mm and a length of 2.1m. The condenser line has an I.D. of 3.86mm and a length of 1.99m. . Ammonia is used as the working fluid. An aluminum heater block with two cartridge heaters is attached to the evaporator to provide a total power up to 1200W. The condenser is cooled by a refrigerator. There are a total of 39 thermocouples to monitor the loop temperatures. A differential pressure transducer is installed across the evaporator and CC to measure the external pressure drop. A metering valve is installed on the vapor line. By adjusting the metering valve, different pressure drop can be imposed upon the loop. A data acquisition system and Labview software are used to monitor and store data. Details of the test setup can be found in Reference 3.

**Testing Methodology**

The purpose of this study is to demonstrate that: 1) the loop operating temperature will change when the external pressure drop is changed while other conditions remain the same; 2) the effect of the pressure drop on the operating temperature is dependent upon the void fraction inside the evaporator core.

Since it is not possible to measure the pressure drop across the meniscus at the liquid/vapor interface formed at the primary wick, only the pressure drop from the vapor line to the liquid line is monitored using a differential pressure transducer, as shown in Figure 3. The external pressure drop can be changed by adjusting the metering valve to different positions. Thus, any *change* in the pressure drop imposed by the metering valve also represents the *change* in the pressure drop across the primary wick. Similarly, the vapor temperature in the evaporator grooves and the CC are not measured directly. Only the wall temperatures are measured by the thermocouples attached to the outer surfaces. As the pressure drop is changed, the heat leak and the CC temperature will change. In other words, the entire study will concentrate on how the *change* in the external pressure drop affect the *change* in the loop saturation temperature.

It is expected that the vapor void fraction inside the evaporator will change with the fluid inventory and the tilt between the evaporator and the CC. Tests were performed with three fluid inventories: 83 grams, 100 grams, and 113 grams. Figure 4 shows the cross sectional dimensions of CC and the evaporator, and the liquid fill levels at the three fluid inventories. The liquid fill level is derived by assuming that the evaporator and CC are both horizontal and that all components except the evaporator and the CC are completely filled with liquid. Since the initial condition was not controlled in this test program, the actual liquid-fill level of the evaporator and CC was not known. However, once the flow circulation is established, a higher fluid inventory will lead to a higher liquid-fill level. Three tilts were used: adverse tilt of –6.35mm (CC below the evaporator), favorable tilt of +6.35mm (CC above the evaporator), and flat (0mm).

Two heat loads were applied: 100W and 200W. At these power levels, the metering valve is effective in generating significant pressure drop only in the last 0.1 percent of its operational range. Although the loop saturation temperature will change more drastically with changes in the pressure drop at lower powers, it was impossible to operate the metering valve and get consistent pressure drops at powers much lower than 100W. All tests were performed in ambient condition with the condenser sink at 283K.

The external pressure drops imposed by the metering valve were somewhat arbitrary. The primary wick can sustain about 32000Pa of pressure drop at the room temperature using ammonia as the working fluid. In the case of 83 grams of fluid inventory, two external pressure drops were imposed: 3400Pa and 7000 Pa, which corresponded to 0.6 meter and 1.2 meters of gravity head. At 100 grams and 113 grams, the loop saturation temperature changed very little when an external pressure drop of 3400Pa was applied. Thus, tests were conducted with external pressure drops of 7000Pa and 20000Pa.

**Tests Results**

Test results are summarized in Table 1. It is clearly seen that the operating temperature increases with an increasing external pressure drop. Moreover, such an effect strongly depends upon the interactions among fluid inventory, tilt, and evaporator power.

At the low fluid inventory of 83 grams, the vapor void fraction inside the evaporator core is large. Consequently, a small increase in the external pressure drop can result in a large heat leak and a large temperature increase. Figures 5 and 6 show the loop temperatures at a heat load of 100W and a tilt of –6.35mm and 0mm, respectively. The CC temperature is represented by TC3 and the evaporator temperature by TC7. The effect of the tilt is more pronounced at the low fluid inventory because a small tilt represents a large percentage change in the vapor void fraction. Figure 7 shows the results at –6.35mm tilt and 200W heat load. The larger flow rate at 200W provides higher liquid subcooling and hence partially compensates for the effect due to the external pressure drop, yielding a smaller increase in the operating temperature for the same external pressure drop. This is evident when comparing Figure 7 to Figure 5.

Figure 5
- 12/7/00, 11:00 – 14:30, (83g/-6.35mm, 100W/280K)

Figure 6
- 12/13/00, 9:00 –11:00 (83g/0mm, 100W/280K)***

Figure 7
- 12/7/00, 14:30-17:30, (83g/-6.35mm, 200W/280K)

As the fluid inventory increases to 100 grams, the vapor void fraction and the heat leak decrease. Consequently, the effect of the external pressure drop on the operating temperature decreases. Initial tests showed little change in the operating temperature as the external pressure drop increased from 250Pa (valve fully open) to 3400Pa. It was then decided to conduct the test at external pressure drops of 7000Pa and 20000Pa. Figures 8 shows the operating temperature as a function of external pressure drop at 100W with a tilt of –6.35mm. The operating temperature increased with an increasing pressure drop, and returned to the previous value at as the pressure drop decreased. At this fluid inventory, the tilt still results in variation of the void fraction, leading to variations in the operating temperature for the same external pressure drop. Table 1 shows that the effect of the external pressure drop on the loop operating temperature is more pronounced at –6.35mm tilt than at +6.35mm tilt, which was expected. However, it is perplexing to see that the 0mm tilt had a larger effect than –6.35mm and +6.35mm. A closer look at the data indicates that when the metering valve was closed in an attempt to get 20000Pa, the valve was closed too much for a short time and the pressure drop exceeded 35000Pa, as shown in Figure 9. Since this was higher than the wick capillary limit, it is suspected that vapor penetrated the wick and changed the void fraction inside the evaporator. This is further evidenced by the subsequent events. As the external pressure drop was reduced, the operating temperature did not return to the previous values at the same pressure drops. Test results at a heat load of 200W indicate there was a large jump in the operating temperature when the external pressure drop was increased from 7000Pa to 20000Pa. It is not clear what caused such a drastic change in the operating temperature. The metering valve had to be closed to the last 0.1percent of its operating range in order to yield 20000Pa. It was suspected that a choked flow might have occurred during the transient.

Figure 8
- 1/24/01, 8:00 to 13:00 (100g/-6.35mm, 100W/280K) (should change to 9:00 – 13:00)

Figure 9
1/22/01, 8:30-13:00 (100g/0mm, 100/+10)

At a fluid inventory of 113 grams, the evaporator core is filled with liquid in all tilts. Thus, the loop operating temperature is less affected by a change in the external pressure drop, as shown in Table 1 for the heat load of 100W. At 200W, a large jump in the operating temperature was seen again as the external pressure drop increased from 7000Pa to 20000Pa.

## Concluding Remarks

When an LHP is ground tested at the spacecraft level, in many cases, the evaporator has to be elevated above the condenser. Thus, an additional pressure drop due to gravity is imposed upon the primary wick. Such an external pressure drop will cause the loop operating temperature to increase even if other conditions remain unchanged. Because the gravity head is absent in space flight, difference in the operating temperature can be expected between the ground and flight tests. Moreover, the difference in the operating temperature between the ground and flight tests is a function of the vapor void fraction in the evaporator core. Both the fluid inventory and the tilt between the evaporator and the CC can affect the void fraction. It is important to keep this effect in mind when comparing the ground and flight results or projecting the flight results from the ground test. Fortunately, he effect of the external pressure drop on the loop operating temperature is pronounced only at low heat loads. This effect diminishes at high hat loads.

## References

1. Maidanik, Y. F., et al., "Heat Transfer Apparatus," United States Patent No. 4515209, 1985.
2. Ku, J., "Operating Characteristics of Loop Heat Pipes," SAE Paper No. 1999-01-2007, 1999.
3. Ku, J., Ottenstein, L., P. Rogers, and K. Cheung, "Low Power Operation in a Loop Heat Pipe," SAE Paper 2001-01-1992, 2001.

| Fluid Inventory (g) | Power (W) | External Pressure (Pa) | Tilt = -6.35mm | Tilt = 0mm | Tilt = +6.35mm |
|---|---|---|---|---|---|
| 83 | 100 | 250 | 296.5 | 294.4 | 291.0 |
| 83 | 100 | 3400 | 298.9 | 295.9 | 291.3 |
| 83 | 100 | 7000 | 302.7 | 298.6 | 292.5 |
| 83 | 200 | 1000 | 294.4 | 294.1 | 293.9 |
| 83 | 200 | 3400 | 294.9 | 294.5 | 294.0 |
| 83 | 200 | 7000 | 297.5 | 294.7 | 294.5 |
| 100 | 100 | 250 | 292.2 | 290.7 | 289.8 |
| 100 | 100 | 7000 | 295.8 | 295.6 | 290.1 |
| 100 | 100 | 20000 | 300.7 | 301.2 | 291.2 |
| 100 | 200 | 1000 | 293.5 | 292.9 | 293.2 |
| 100 | 200 | 7000 | 293.7 | 293.4 | 293.2 |
| 100 | 200 | 20000 | 302.7 | 300.7 | 299.4 |
| 113 | 100 | 250 | 291.6 | 291.9 | 291.6 |
| 113 | 100 | 7000 | 292.4 | 292.2 | 292.0 |
| 113 | 100 | 20000 | 293.5 | 293.4 | 294.3 |
| 113 | 200 | 1000 | 296.1 | 295.6 | 295.2 |
| 113 | 200 | 7000 | 300.5 | 297.9 | 295.3 |
| 113 | 200 | 20000 | 304.5 | 302.7 | 298.2 |

Table 1. Loop Operating Temperature

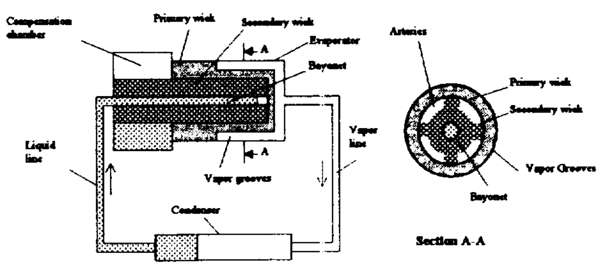

Figure 1 Schematic of an LHP

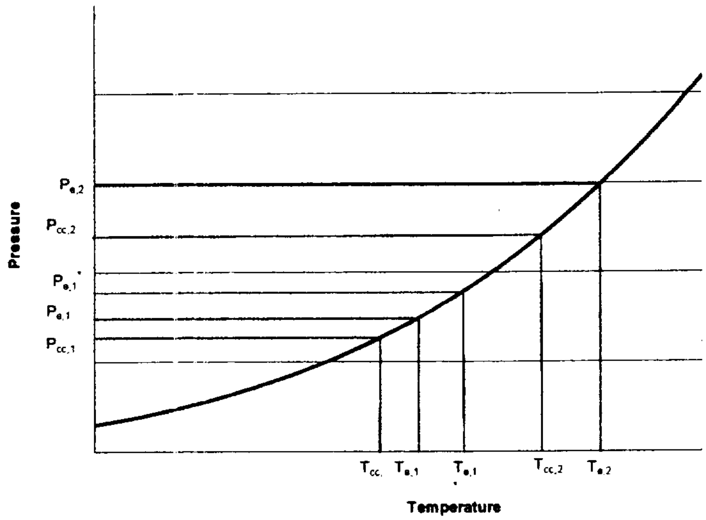


Figure 2

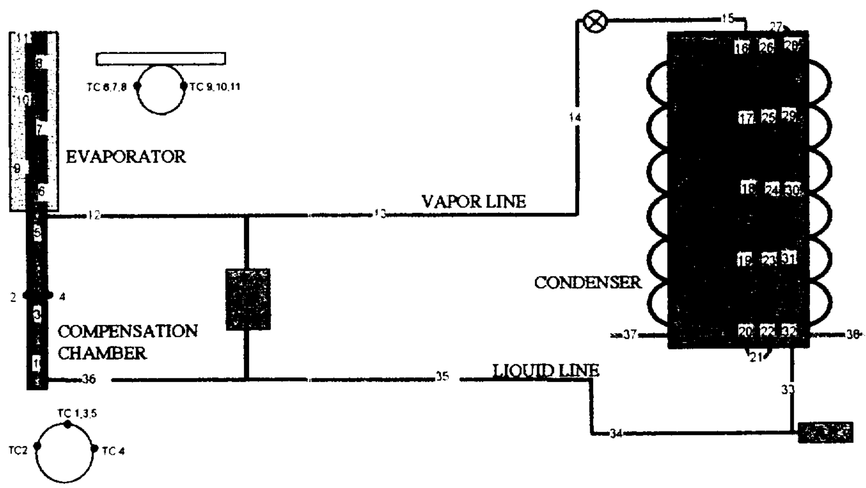


Figure 3 Schematic of the LHP Test Loop

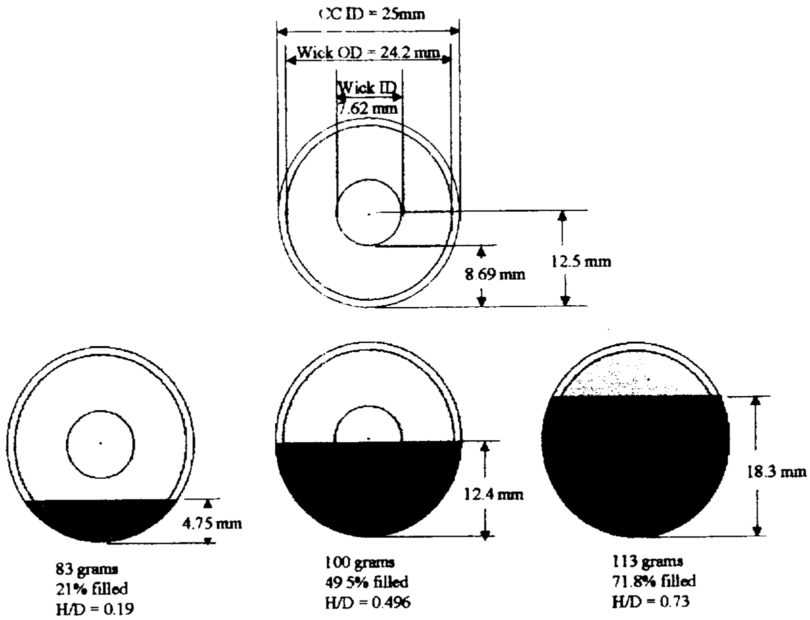


Figure 4 Liquid Fill Level in CC and Evaporator

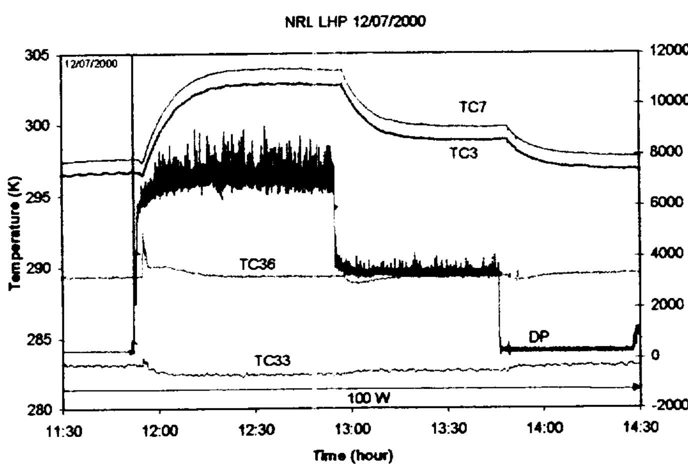


Figure 5. Loop Operating Temperature as a Function of External Pressure Drop (83g/-6.35mm/100W)

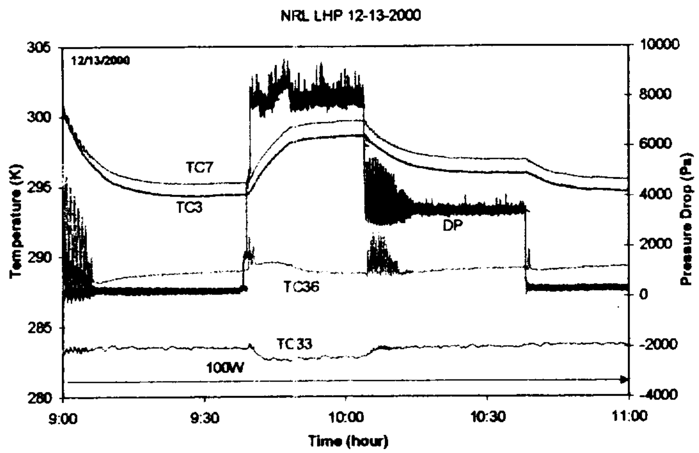


Figure 6. Loop Operating Temperature as a Function of External Pressure Drop (83g/0mm/100W)

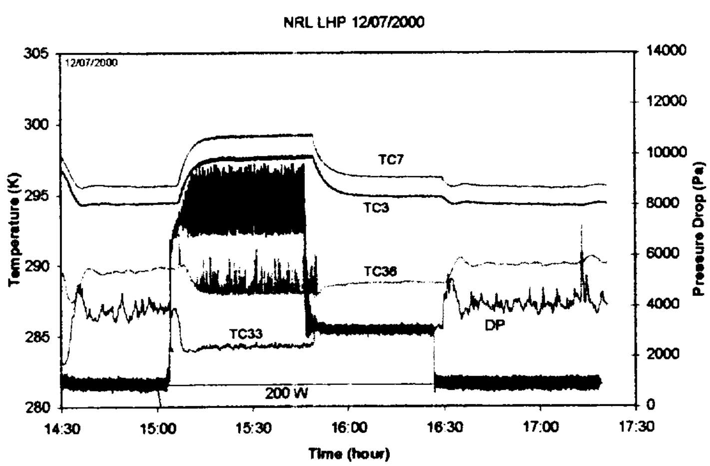


Figure 7. Loop Operating Temperature as a Function of External Pressure Drop (83g/-6.35mm/200W)

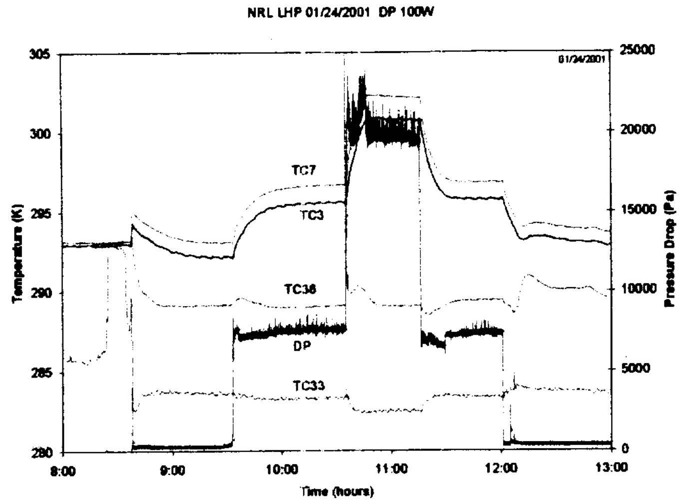


Figure 8. Loop Operating Temperature as a Function of External Pressure Drop (100g/-6.35mm/100W)

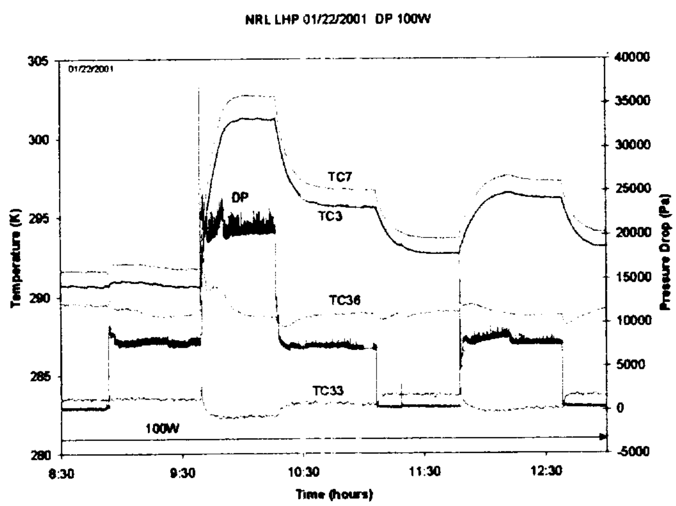


Figure 9. Loop Operating Temperature as a Function of External Pressure Drop (100g/0mm/100W)